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DAMPING BEHAVIOR OF DEHAVILLAND STEM BOOMS

by R. E. Predmore, C. L. Staugaitis, and J. E. Jellison

Goddard Space Flight Center

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ABSTRACT

The results of a program aimed at investigating the vibration frequency and damping behavior of DeHavilland Booms are presented. The program consisted of two phases: (1) A study of the theory of damping mechanisms in metals and alloy systems to predict the results of internal friction and structural damping; and (2) Experiments to measure vibration decay rates under vacuum conditions using strain gage techniques compared with photographic techniques. Damping behavior is discussed in terms of boom orientations, boom length, and the mechanism responsible for the boom dynamics observed. The magnitude of damping was predicted and found in the DeHavilland Boom to be small for both the electroplated and bare conditions. The principal damping mechanism of the DeHavilland STEM is the frictional damping between the overlapping surfaces of the boom. Electroplated silver was found to reduce the damping because it lubricated the sliding interfaces of the overlap.

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INTRODUCTION

The DeHavilland STEM* concept has been employed for a variety of applications including antennas and gravity gradient stabilization. While data have been accumulated on the static and thermal properties of this particular tubular element, there exists at the present time a scarcity of meaningful information on the dynamic behavior, which is essential for spacecraft system design and prediction of service performance. Consequently, a test program was initiated with the primary aim of investigating the vibration frequency and damping behavior of 43 foot long DeHavilland booms. The decaying oscillations were monitored by strain gage instrumentation of the boom root and compared with the results obtained by photographic techniques.

As part of a broad GSFC materials investigation involving DeHavilland type (STEM) booms, this effort has been undertaken to determine the amount and the mechanism of vibrational damping developed by this particular boom concept. The program is divided into two phases: (1) A study of the theory of damping mechanisms encountered in metals and alloy systems to predict the contributions expected from the various sources of internal friction and structural damping in the booms; and (2) an experimental program to measure the vibration decay rates under vacuum conditions using strain gage techniques. Damping of each boom is discussed in terms of boom orientations, boom length, and mechanism responsible for the boom dynamics observed.

THEORY OF DAMPING MECHANISMS

Prior to undertaking the experimental program, several potential damping mechanisms were evaluated in the light of the BeCu alloy boom material, and the environmental conditions expected in the proposed application (References 1 and 2).

To distinguish between damping phenomena directly attributable to structural factors and those arising from material properties inherent in multi-layer booms, ancillary tests were conducted to

*STEM (Stored-Tubular-Extendible-Member). A DeHavilland Aircraft of Canada Ltd. concept permitting storage of long booms in locations of limited capacity.

evaluate the damping properties of selected plating materials. Results of these analyses are summarized below.

Phenomenological Damping Mechanisms

Stress Induced Ordering

Interstitial Ordering - The interstitials N_2 and C induce damping in iron at about 70° but the effect disappears when the residuals are removed. BeCu alloy is free of interstitials in the copper crystal matrix and thus damping by interstitial ordering cannot develop.

Precipitates of carbon or nitrogen and the matrix elements formed from a supersaturated solid solution can promote damping; but, in the absence of interstitials, this type of damping mechanism cannot operate.

Substitutional Solution Ordering

Damping by stress wave induced, substitutional solution ordering has been observed in such FCC alloys as Zn in a solution of Cu and Zn in Ag. The mechanism of substitutional solution damping is observed only in alloys of 10% or more solute. BeCu alloy contains 5% or less solute; therefore, damping by stress induced redistribution of solute atoms is considered insignificant. Furthermore, this type of damping is observed in the Ag-Zn system at 200°C, which is well above the service temperature anticipated for the BeCu alloy.

Damping caused by stress wave induced disordering from an initially ordered substitutional solution will not occur because BeCu alloy is not initially ordered.

Movement of Twin Interface

Damping resulting from twin boundaries or martensitic transformation has been observed at ambient temperatures in CuMn alloy. BeCu alloy does not develop twins nor does it undergo a martensitic transformation; therefore, damping by this mechanism is not possible.

Grain Boundary Viscosity

Damping occurs in copper alloys by shear relaxation across incoherent grain boundaries at temperatures above 200°C. Significant damping by this process is not possible because the BeCu alloy will not reach this temperature in space.

Plastic Damping or Dislocation Damping

Dislocation densities corresponding to cold reductions of about 5 percent induce damping at ambient temperatures with frequency ranges about 2 kilocycles. Since BeCu alloy booms will not be subjected to vibration frequencies as high as 2 kilocycles, dislocation damping, even for small cold reductions, will not be a contributing factor.

Koster damping, the interaction of precipitates with dislocation fields, occurs in highly cold-worked metals subjected to low levels of stress and low frequencies within a temperature range of -100°C to 0°C. The highly cold-worked BeCu alloy for gravity gradient booms can experience Koster dislocation damping approaching $\delta = 3.8 \times 10^{-4}$.

$$\delta = \frac{1}{n} \ln \frac{x_1}{x_m} ,$$

where δ = logarithmic decrement, n = number of cycles, x_1 = amplitude of the 1^{st} cycle, and x_m = amplitude of the n^{th} cycle.

As the temperature is raised in a heavily cold-worked metal, dislocation viscosity damping becomes significant; however, this will not appear in BeCu alloys in the intended space application because the service temperatures anticipated are not high enough.

Thermoelastic Internal Friction

Thermoelastic damping at low frequencies is small because very little heat is generated in the compression stress areas and little cooling occurs in the tension stress areas. BeCu alloy booms used in space normally operate at low frequencies, thus precluding the development of significant thermoelastic damping.

Microscopic thermal currents result in damping at high frequencies because of elastic anisotropy from grain to grain. This type of damping is not expected to account for a significant proportion of the total damping occurring in the gravity gradient rod because, as stated previously, the frequency of the boom is too low.

Magnetoelastic Internal Friction

The coupling between magnetic and mechanical properties of a ferromagnetic material provides a source of internal friction; however, in the case of BeCu, this is not a factor since it is virtually nonmagnetic.

Scattering

At high frequencies (1 MHz $^+$), phases scatter the periodic input stress waves to break up the wave fronts in an alloy and cause damping. BeCu alloy has second phases of cobalt-beryllide which are dispersed through the matrix. As indicated previously, these phases, while having the potential to promote damping, will not because the anticipated service frequency of the gravity gradient booms will be too low.

Structural Damping Mechanisms

Damping of Electroplated Wire

Essentially, the damping developed by the plated booms represents the contribution by the respective metallic films, together with the frictional energy generated by the overlapping surfaces during the boom's bending and twisting excursions. In an attempt to obtain a measure of the damping inherent in the metallic coatings, a separate but related study was carried out. This was prior to the fabrication of the booms, and involved a variety of electroplated films deposited on 0.018 inch diameter, high-carbon steel wire (music wire). The torsional damping properties of Ni, Ag, Zn, Cu, Sn, Cd, and solder were measured using 18 inches of music wire suspended from a frame to form a pendulum with a weight fastened at its tip. The decay rates for a given initial twist angle were recorded and graphically summarized in Figure 1. It is readily evident from these results that cadmium does promote rapid damping of the torsional pendulum. Additional tests over a range of thickness of the plating showed the same relative damping constants as shown in Figure 1, although the constants increased with increasing thickness for a particular plating. Accordingly, booms having a cadmium layer were included in the test program.

Friction Between the Overlap Surfaces

During the oscillations of the boom as indicated in the preceding section, the overlapping edges rub together, and mechanical energy is dissipated by the contacting surfaces. When the boom is silver plated, the silver acts as a lubricant between the surfaces thus reducing the magnitude of the friction and, consequently, boom damping. When cadmium is plated on the surface of the boom, it will damp the stress waves if they are transmitted into the cadmium layer. Since it has a low yield strength, it will plastically deform, thereby dissipating the mechanical energy. However, the bare boom could not be expected to have appreciable structural damping unless the overlap pressure was large.

EXPERIMENTAL PROCEDURE

Materials

A description of the test booms investigated is presented in Table 1. All were fabricated by DeHavilland Aircraft of Canada Ltd. from a .002 inch thick strip of copper-beryllium alloy 25, in the fully aged (XHMS) condition. A DH proprietary method of brush plating was employed to apply the metallic layers of silver and cadmium to the thicknesses indicated. However, a metallographic

Table 1

Description of Test Booms.

Specimen Number	Nominal Boom Diameter (inches)	Boom Length (feet)	Basic Alloy (BeCu)	Thermal-Mechanical Alloy Treatment	Surface Condition	Nominal Plate Thickness (inches)
1	0.50	43	25	XHMS	Silver	0.0002
2	0.50	43	25	XHMS	Bare	
3	0.50	43	25	XHMS	Silver-Cadmium	0.0001 Ag
4*	0.50	43	25	XHMS	Silver-Cadmium	0.0001 Ag
5	0.50	43	25	XHMS	Silver-Cadmium	0.0001 Ag

*This specimen was inadvertently damaged and omitted from the test schedule.

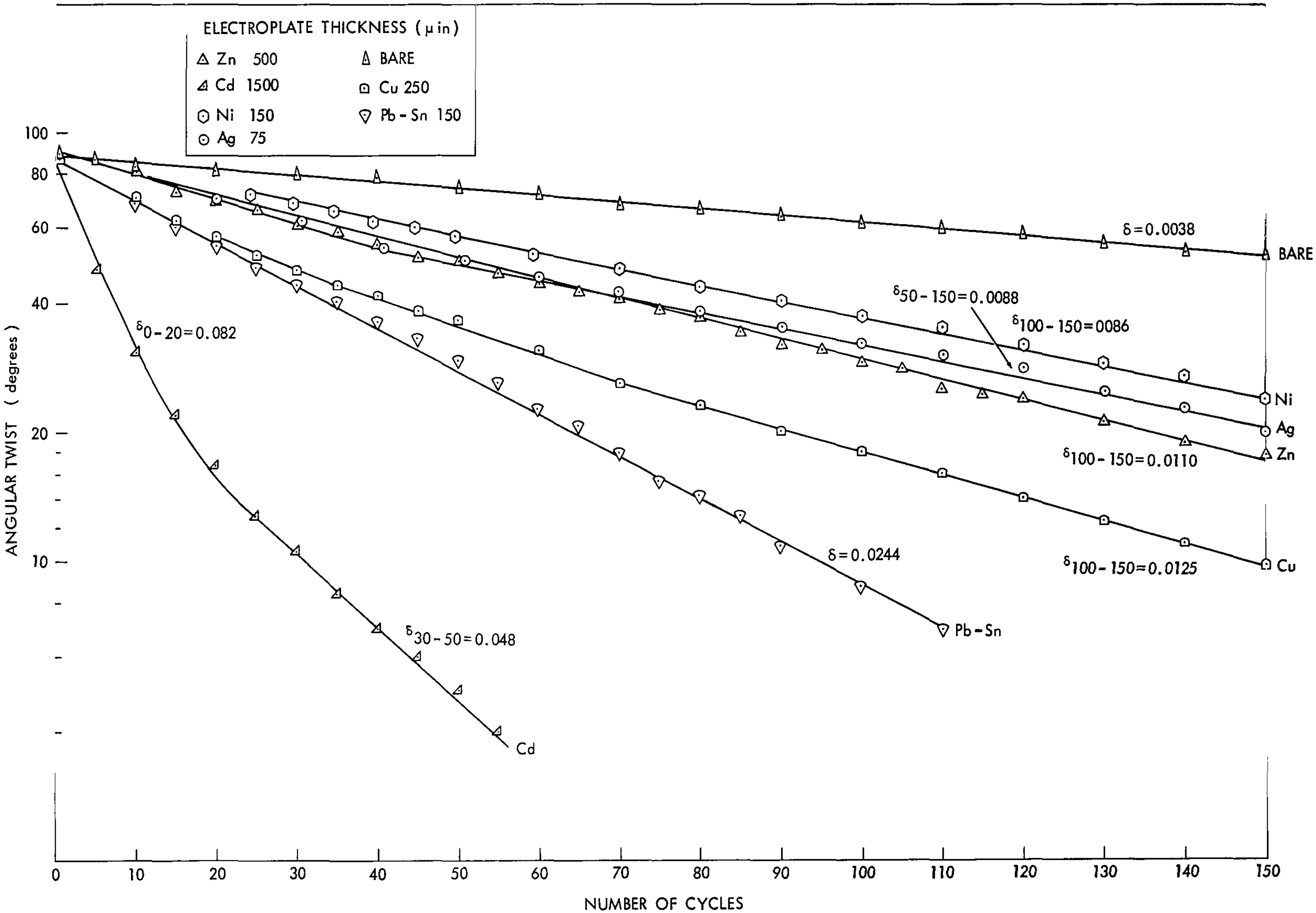

Figure 1—Torsional vibration damping of electroplated music wire 0.018 inch diameter and 18 inches long.

examination of samples removed from Test Booms 3 and 5* revealed that the cadmium layer of these multi-layered booms was discontinuous and non-uniform, with less than 5% of the surface area covered. Discrete deposits ranging from 100 micro-inches to 150 micro-inches thick were observed along sections of the boom, but much of the periphery appeared to be devoid of any cadmium layer. The silver deposits were uniform and about 75 micro-inches thick.

Test Procedure

The 43 foot long test tapes were deployed using a specially constructed DH extension mechanism designed to simulate service operation. Due to the relatively long boom lengths, the tests were performed in the large GSFC Vacuum Facility. During extension, the flat strip is formed into a tubular configuration with a nominal diameter of approximately 0.50 inch. At this point the element is supported by a circular teflon guide. Booms 1, 2, and 3 were deployed through a 5/8 inch diameter guide while a 1/2 inch diameter guide was employed for Boom 5.

This modification was incorporated to promote greater rigidization at the clamped end of the tubular element. In essence this feature introduces a pseudo clamped condition which is analogous to the constraints offered by the mandrel employed in the actual deployment system.

The test setup was specifically designed to permit varying the boom seam orientation by remote control. Boom deflection was achieved by employing a traveling arm device which captured the boom approximately 3 feet from its tip and displaced it 30 inches from the vertical. Upon command, the boom was subsequently released and set into oscillation in a vacuum of about 2×10^{-2} torr. The boom's motion is the response of a clamped-free cantilever beam forced by the boom's elastic stiffness and gravity.

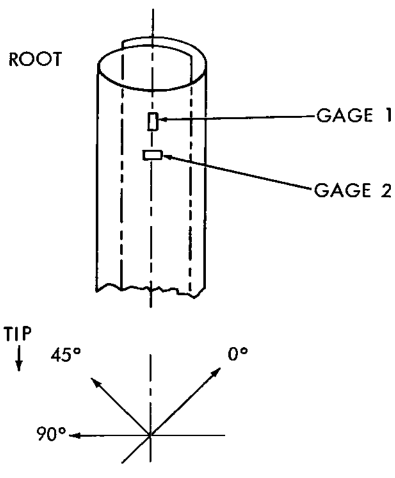

Figure 2—Location of strain gages on Boom 1.

The booms were loaded in a plane perpendicular to the boom axis and forming an angle of 0°, 45°, and 90° with a plane defined by the boom axis and the bisector of the overlap. The boom damping constant was approximated from log plots of strain amplitude response vs. time curves. The damping constants were also approximated from photographs of the tip motions†.

The strain gage instrumentation employed on Boom 1 consisted of two Budd C-9-141B foil strain gages at the base of the boom approximately 1/2 inch below the teflon support guide opposite the overlap (Figure 2). Twelve volts were applied across the gages in a series electrical circuit and the resulting output observed on a 1508 Visicorder‡. To further increase gage response (sensitivity), Boom 2 was instrumented with three

*GSFC Memo to C. Staugaitis from R. Predmore, "Materials R&D Metallographic Examination of DeHavilland Multiplated (Ag-Cd) BeCu Booms," April 26, 1965.

†GSFC Memo from D. J. Hershfeld, "RAE Boom Damping Study in the Dynamic Test Chamber," August 17, 1965.

‡GSFC Memo to C. Staugaitis from R. Predmore, "Materials R&D Progress Report on Damping of 43 feet RAE Booms," February 17, 1965.

silicon semi-conductor gages (BLH-7NB3-18-12-S9) in the same locations. In this case all three were placed opposite the overlap and oriented parallel, perpendicular, and at 45° to the bisector of the overlap respectively and subjected individually to 1-1/2 volts (Figure 3). Booms 3 and 5 were similarly instrumented with semi-conductor gages, but their number and orientation were substantially changed as shown in Figures 4 and 5 respectively.

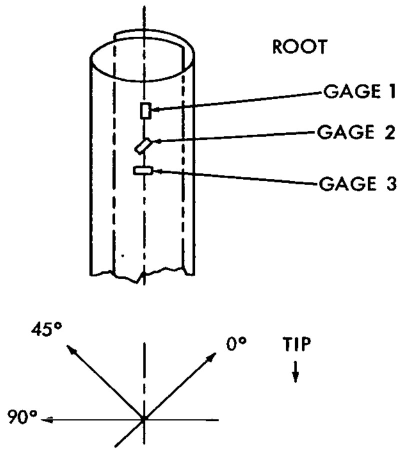

Figure 3—Location of strain gages on Boom 2.

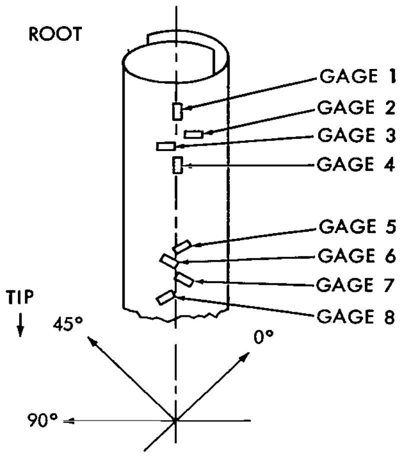

Figure 4—Location of strain gages on Boom 3.

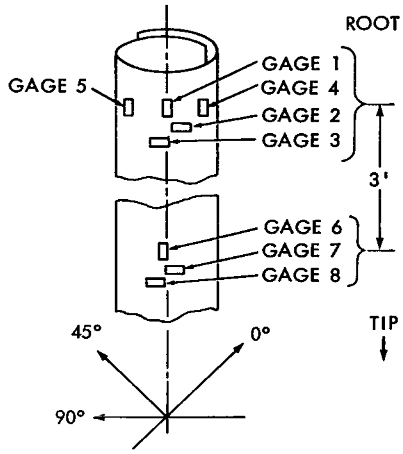

Figure 5—Location of strain gages on Boom 5.

EXPERIMENTAL RESULTS

The strain response vs. time is plotted on a log scale in order to compare each cycle of the test. If the amplitudes decay in an exponential fashion, the graph will be linear and the slope will determine the logarithmic decrement. If the slope is constant, the damping is constant and independent of amplitude. If the plot is not a straight line, damping depends upon the amplitude of the various cycles throughout the test. The strain response from Booms 1 and 2 was recorded from the gage mounted parallel to the boom axis. The maximum cyclic stresses recorded at the base of the various booms were about 3000 psi and decreased sharply with the number of cycles and with the distance from the root of the boom.

The strain gages were positioned on Boom 3 as shown in Figure 4 to see if the stress-strain behavior compared with that of a thin wall cylinder. The clamped-free cantilever beam with the tip deflected has the greatest moment and the greatest stresses at the base. Strain Gages 1 and 4 were positioned to record the bending stress as the boom oscillated in the 0° direction. Strain Gage 4 was expected to produce the same response as Gage 1. Strain Gages 2 and 3 were positioned to record stresses resulting from changes in the boom diameter. Strain Gages 5, 6, 7, and 8 were positioned to record the strain between Gages 1 and 2, etc.

The strain decay curve measured from Gage 4 on Boom 3 was virtually identical to the strain decay from Gage 1. The cyclic strain response recorded from Gage 2 was 180° out of phase from the strain response recorded from Gage 3. The strain responses recorded from Strain Gages 5, 6, 7, and 8 were about proportional to the sum of the strain responses from Gages 1 and 2, 1 and 3, 2 and 4, and 3 and 4, respectively.

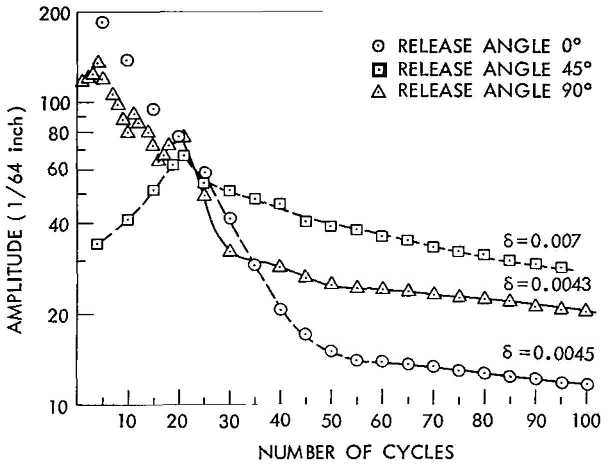

Figure 6—Strain decay and logarithmic decrements measured from Gage 1 on Boom 3.

The strain responses recorded from Gage 1 on Boom 3 are plotted in Figure 6 for cases where the boom was set into oscillation in the 0°, 45° and 90° planes. The analysis* of the strain response from Boom 3 led to the preliminary conclusions that the mechanics of the boom were determined by assuming that the boom was warping, bending, and shearing, simultaneously. In order to learn more about the warping, bending, and shearing deformations, eight strain gages were arranged as shown in Figure 5. Measurements of cyclic stressing as a function of boom length and damping were recorded. The response from Gages 6, 7, and 8 on Boom 5 was about half as large as the response from the gages at the root of the boom; thus the maximum cyclic stresses were in the range 1500 to 2000 psi. Strain Gage 6 was rotated approximately 130° about the boom axis from Gage 1. The 130° rotation is a measure of the inherent twist in the boom caused by the fabrication of the boom itself. Strain Gages 1, 4, and 5 were mounted parallel to the boom axis and 80° apart. A comparison of their outputs was expected to show longitudinal stresses from cantilever bending and shear stresses from the boom's motion. Gages 2 and 3 were expected to measure shear stresses equal to that measured in the vertical gage, and boom diameter changes caused by cantilever bending. Gages 7 and 8 were expected to record the same strains as Gages 2 and 3.

The strain response recorded from Gage 5 completely decayed after 30 cycles and built up again after 45 cycles. The photograph of the tip deflection showed that the neutral axis of the boom rotated about the boom axis while it was swinging. The attenuated oscillation recorded from Gage 5 between the 30th and 45th cycles was caused by the neutral axis passing through the gage between the 30th and 45th cycle. Because the neutral axis continuously rotates, the amplitude decay is governed both by the rotation of the neutral axis and the damping of the boom. Thus, the decreasing amplitude with time recorded from the strain gages must be separated into actual and apparent damping of the vibration energy. The response from Gages 1, 4 and 5 and the phase angles between them could be predicted within a fair amount of accuracy by considering the shear strain and the longitudinal strain from cantilever bending. The strain decay curves recorded from Gage 1 on Boom 5 are shown in Figure 7.

*GSFC Memo from Harold P. Frisch, "Frictional Damping Inherent in the RAE Boom, a Means for Damping-out Large Amplitude Vibration," July 19, 1965.

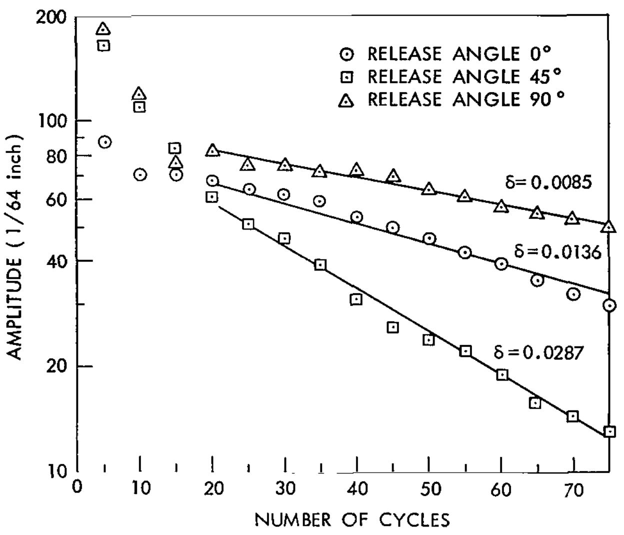

Figure 7—Strain decay and logarithmic decrements measured from Gage 1 on Boom 5.

To observe the change in boom damping as a function of boom length, Boom 5 was retracted to a length of 28 feet. The strain gages were positioned on the boom as shown in Figure 2. The strain decay curves and the logarithmic decrements recorded from Gage 1 are shown in Figure 8.

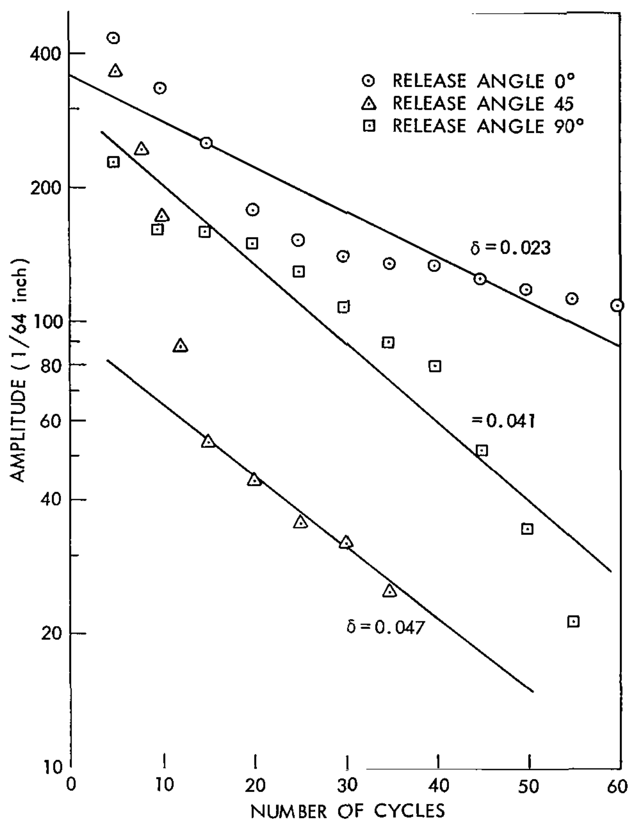

Figure 8—Strain decay and logarithmic decrements measured from Gage 1 on 28 foot Boom 5.

DISCUSSION

The strain decay behavior and the logarithmic decrements measured from Gage 1 in Figures 6, 7, 8, and 9 on Booms 1, 2, 3, and 5 are about the same as the strain decay curves and logarithmic decrements from the photographic measurements.

Table 2 gives the logarithmic decrements measured by Gage 1 and by the photographic technique. The total damping is caused in part by viscous internal friction damping mechanisms and in part by nonviscous friction damping. However, since in this case the total damping is small, the viscous and nonviscous components of damping have been combined and are represented by the viscous damping constant δ (logarithmic decrement).

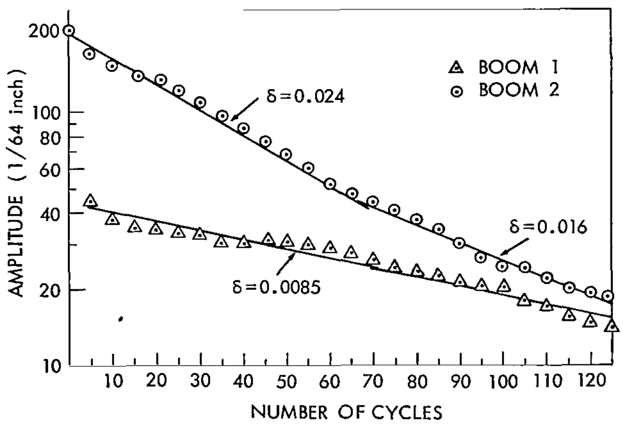

Figure 9—Strain decay and logarithmic decrements measured from Gage 1 on Booms 1 and 2.

Table 2

Logarithmic Decrements for Booms 1, 2, 3, and 5.

Test Booms	Angle of Pickup and Swing	Logarithmic Decrement	
		Strain Gage 1	Photograhpic
1 Ag plated BeCu (43 feet)	90°	0.0085	0.0081
2 BeCu (43 feet)	90°	0.024	0.0276
3 Ag-Cd plated BeCu (43 feet)	0°	0.0045	0.0066
	45°	0.0043	0.0071
	90°	0.007	0.0074
5 Ag-Cd plated BeCu (43 feet)	0°	0.0136	0.0267
	45°	0.0287	0.0287
	90°	0.0085	0.0083
5 Ag-Cd plated BeCu (28 feet)	0°	0.041	0.0470
	45°	0.047	0.0470
	90°	0.023	0.0470

The strain decay curves recorded from the other gages on Booms 2, 3, and 5 do not agree with Gage 1. Because of the agreement between photographic data and Gage 1 data, it is concluded that the strain recorded by Gage 1 is principally caused by the longitudinal stresses resulting from cantilever deflection of the boom. Thus a strain gage in this position can be used to measure the strain decay of the boom. The warping of the boom, the shear rotation of the boom, and the cantilever bending of the boom are responsible for the strains recorded in Gages 2, 3, 5, 6, 7, and 8 on Boom 3 and Gages 2, 3, 4, 5, 6, 7, and 8 on Boom 5. Therefore the strain decay curves measured from the above gages will not agree with the photographic data or Gage 1.

The damping of Booms 3 and 5 is much higher during the early cycles. Initially the boom contains torsional vibration and transverse vibration at frequencies higher than the fundamental frequency. These motions produce large amounts of overlap slippage. The friction between the overlap surfaces damps these high frequency vibrations quickly.

A comparison of the damping behavior of Booms 3 and 5 illustrates the effect of clamping conditions at the root of the boom. Boom material and test conditions were the same in each case except for the diameter of the teflon guide at the deployment device. Decreasing the diameter of the guide from 5/8 inch (Boom 3) to 1/2 inch (Boom 5) has greatly enhanced the damping. When the boom is confined by a 5/8 inch guide, the outside of the boom does not touch the guide during the center portion of the swing. The boom is thus free to pivot on the open portion of the boom immediately adjacent to the deployment device. This open, flat ribbon portion acts like a hinge at the root of the boom, decreasing the rigidity at the base, thus reducing the overlap interfacial slip and friction. Consequently, the damping capacity of the boom is lowered. When the diameter of the guide is reduced to 1/2 inch, the perimeter of the boom is confined by the guide, creating more overlap interfacial movement near the root of the boom and thus more damping.

The comparison of the damping of silver plated Boom 1 ($\delta = 0.0085$) with the bare Boom 2 ($\delta = 0.024$) in Table 2 shows that the silver plated boom has much less damping than the bare boom. The silver acts as a lubricant between the sliding overlap surfaces and consequently reduces the friction and damping. It is noted that the damping of the silver plated wire ($\delta = 0.0088$) is nearly equal to the damping of the silver plated Boom 1 ($\delta = 0.0085$). There seems to be no apparent reason for this equality, and it is regarded as a coincidence.

A further increase in damping is observed when the 43 foot Boom 5 is withdrawn to a length of 28 feet. For any length, the bending stresses and high frequency vibrations are concentrated within approximately the first 15 feet from the base of the boom, thus indicating that most of the energy is dissipated in this region. For a 28 foot and a 43 foot boom with equal maximum bending moments, the 43 foot boom has a larger energy per cycle. Friction damping removes an equal amount of energy per cycle from each boom and thus the 28 foot boom is damped more rapidly.

The fact that boom damping increases with decreasing guide size and decreasing boom length shows the clamping condition at the root of the boom is an important factor in the tests. It also indicates that most of the damping is caused by friction between the overlapping surfaces of the boom.

The damping of each boom was dependent upon the angle between the diameter through the center of the overlap and the direction from which the boom was initially released (Table 2) or the release angle (Figure 6). The release angle varies the twist imparted to the boom which changes the torsional vibration imparted to the boom. The clamping rigidity at the base of the boom depends upon the relation of the release angle and the orientation of the flat part of the boom adjacent to the deployment device. As a consequence, the high rate of torsion vibration damping and the variation of rigidity at the root are directly related to the boom release angle.

CONCLUSIONS

The results of the investigation permit the following conclusions:

1. The magnitude of damping was predicted and found in the DeHavilland boom to be small for both the plated and unplated conditions.
2. It is evident that damping depends not only on the boom itself, but also on its length, plane of oscillation relative to the overlap, and the manner in which it is supported at the base.
3. Photographic and strain gage measurements show good agreement in their description of the dynamic response of the DeHavilland boom.
4. A qualitative understanding of the dynamics of the boom was obtained from dynamic response data recorded by the strain gages and the photographs.
5. The principal damping mechanism of the STEM booms is frictional damping between the overlapping sides of the boom.
6. Electroplated silver was found to reduce the damping because it lubricated the sliding interfaces of the overlap.

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National Aeronautics and Space Administration
Greenbelt, Maryland, June 3, 1966
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